

AN INVESTIGATION OF LUBRICATING OIL
FILM BREAKDOWN PRESSURES UNDER
STEADY AND INTERMITTENT LOADING

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D. N. CONE
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THE PENNSYLVANIA STATE COLLEGE
Department of Mechanical Engineering

AN INVESTIGATION OF LUBRICATING OIL FILM BREAKDOWN
PREDICTED UNDER STEADY AND INTermittENT LOADING

A Thesis
By
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and
Lieutenant (junior grade) ARTHUR D. BARNES, U. S. NAVY

Submitted in partial fulfillment
for the degree of
MASTER OF SCIENCE

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Approved: May 19, 1932
Louis J. Bradford
Professor of Machine Design

thesis
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bogboard fairlead

Introduction

The investigation of breakdown pressures of lubricating oil films was an attempt to establish a definite relation between the effects of steady and intermittent loads on a bearing. The work was extended to include the correlation of as many of the variables which enter into the establishing and maintaining of an oil film as was possible. Such an investigation had been instituted in 1931 by Lieutenant R. L. Blue, U. S. Navy, and Lieutenant (junior grade) R. L. Ewart, U. S. Navy. Credit is due them for their pioneering work in the construction of the basic machine and initiating the test. Various recommendations by Lieutenants Blue and Ewart for modifications in the machine and procedure have been carried out insofar as time and funds have permitted.

It was necessary to obtain additional equipment and replace some of the original which was not available for our use during this current year. Considerable time was spent in locating and obtaining this equipment, in manufacturing various parts of the apparatus, and in making the various modifications.

Description of Apparatus

The machine proper for testing the breakdown pressure of oil films consists primarily of a test bearing and shaft, a loading mechanism by which the load is applied to the test bearing, a means of varying the magnitude of the load, a mechanical method of alternately applying and releasing the load, and a means of furnishing a continuous and constant supply of lubricating oil to the test bearing.

TEST BEARING.

The test bearing is a bronze shell with a babbitt lining approximately $.125^{\prime \prime}$ in thickness. The shell was originally $4 \frac{1}{4}^{\prime \prime}$ in diameter and $2^{\prime \prime}$ in length, but has been somewhat flattened on the top to provide a convenient surface for application of the load. The overall diameter has been further increased by the construction of a water-jacket of sheet copper material, having a water space of $1/2^{\prime \prime}$. The jacket extends around the two sides of the circumference as shown in Figure 6. It was not practical to extend the jacket completely around the circumference due to the necessity of applying the load to the bearing on the top, and of introducing the lubricating oil into the bearing at the bottom. A thermometer well was drilled radially into the shell, and tangent to and in

contact with the babbitt as far as practicable. The babbitt was turned to a nominal diameter of 1.998" and reamed to give a .006" diametral clearance on the test shaft. An oil groove 1 3/4" long, 3/16" wide, and 1/8" deep was cut axially along the bottom of the bearing at the point where the lubricating oil is introduced.

TEST SHAFT.

The test shaft is of case-hardened steel, ground and polished to a diameter of 2.0005" along its length. It is 21" long, both ends extending beyond its support bearings sufficiently to permit the installation of a pulley on one side and the use of a hand tachometer on the other. The shaft is not restrained axially, but is free to find its own running position in the bearing. It is belt-driven by a 5 H.P., 120 volt, shunt wound, direct current, motor, speed control of which is obtained by means of two portable lamp banks in the field circuit.

LOADING MECHANISM.

The loading mechanism consists of several parts, namely, a load bar, four 3/4" steel tie-rods, and a strong back which receives the load transmitted by the tie-rods from the load bar. The strong back loads the bearing which is located at its central point. An attempt was made to avoid eccentricity of loading by using a 3/8" steel ball between the strong back and test bearing. It was hoped

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Съдържание

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използвани за издаването на документа. Той съдържа:
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that the "point loading" would not cause canting of the bearing on the test shaft with a possible resultant local breakdown of the oil film. In addition to the ball, three $1/8"$ strips of bakelite were inserted between the strong back and bearing for electric insulation. Two stud bolts from the bearing extend through the strong back acting as its vertical guides, and also serve to prevent the test bearing from rotating with the shaft.

The strong back is made of mild steel $4" \times 2" \times 13"$ and is sufficiently heavy so that its deflection due to center loading is negligible. It is drilled at each end to receive the tie-rods and transmitted load from the load bar. It is also drilled to permit the passage of two oil cup feeds to the test shaft support bearings and a rubber hose jumper-connection between the water jackets on the two sides of the test bearing.

The load bar is of mild steel material $2" \times 1" \times 14"$. It was annealed at 1475°F . in the electric furnace for ten minutes and then drawn, after an oil quench, at 650°F . for thirty minutes to obtain sufficient strength to carry a center load of 5000 pounds without taking a permanent set. It was next calibrated in a hand-operated, tension-testing, machine to obtain a load-deflection curve. A dial indicator was suitably mounted on a flat plate directly under the point of loading to test deflections

of the bar under the various loadings could be accurately measured. From these deflections and the applied load, as obtained from the testing machine, a load deflection curve was constructed. The curve was modified to include the weight of the loading mechanism, which was 58 1/2 pounds, so that the actual bearing pressure in pounds per square inch of projected area could be obtained for any deflection. This curve is shown in Figure 4. Cup seats at each end of the load bar transmit the load to the tie-rods.

The point of loading is the head of a 3/8" hexagonal bolt secured on the bottom of the lever bar. The latter is a specially machined steel shape from original mild steel stock 2" x 2 1/2" x 1/2". It is fitted at one end with a 2" case-hardened roller follower which remains in contact with a rotating cam. The lever is supported by two springs with sufficient tension to insure contact between cam and follower. It is pivoted at the other end on a piece of 1" round steel stock so that a ratio of the distance from the pivot to the roller to the distance from the pivot to the load bar is 5 to 1. In this way the load on the load bar is five times the pressure which the cam exerts on the follower.

The cam is case-hardened, medium carbon steel. It is 2" wide and is designed to give a lift of 1/4" over

an arc of 180 degrees. The car is rotated through a Chevrolet transmission by a 2 H.P., 120 volt, shunt wound, direct current motor. The latter is fitted with a field rheostat for speed control. As the transmission allows speed ratios of 3:1, 1 1/2:1, and 1:1, a wide range of speed for the car is available.

ELECTRICAL CIRCUIT.

The closing of an electrical circuit by oil film breakdown determines the point of breakdown conditions. The circuit consists of four dry cells connected in parallel and in series with a small 6 volt test lamp, an ammeter and a knife switch. One lead from this apparatus is taken to one of the test bearing saddle bolts above the strong back. The other goes to a suitably mounted copper wire which comes in contact with the rotating test shaft. As long as an oil film is maintained between the shaft and test bearing, the electrical circuit is interrupted and no current can flow. At the instant of oil film breakdown, the circuit is complete, the lamp lights, and the ammeter needle is deflected.

LUBRICATING OIL SUPPLY.

The lubricating oil supply is from a tank located 26 feet above the machine. The oil, which is previously centrifuged, is received at the bearing through a filter at a pressure of 9.5 pounds per square inch. Time per-

и супервайзингом. Важно помнить, что в концепции А. А. Томпсона и А. А. Гарднера (1995) терапевтическое взаимодействие между клиентом и терапевтом не является самоцелью, а средством достижения целей, определенных клиентом. Важно помнить, что терапевт несет ответственность за то, чтобы клиент не терял интерес к процессу и не терял веру в то, что он может изменить свою жизнь. Терапевт несет ответственность за то, чтобы клиент не терял интерес к процессу и не терял веру в то, что он может изменить свою жизнь.

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Несмотря на то что терапевт несет ответственность за то, чтобы клиент не терял интерес к процессу и не терял веру в то, что он может изменить свою жизнь, это не означает, что терапевт несет ответственность за то, чтобы клиент не терял интерес к процессу и не терял веру в то, что он может изменить свою жизнь. Терапевт несет ответственность за то, чтобы клиент не терял интерес к процессу и не терял веру в то, что он может изменить свою жизнь.

mitted the test to be made with only one lubricant, the characteristics of which are here given.

TEXACO REGAL "C" OIL

Gravity 20.1

Flash point 365°F.

(Cleveland Open Cup Method)

Fire point 415°F.

Viscosity in Saybolt Seconds

100°F. 324 seconds

130°F. 137 seconds

75°F. 1090 seconds

Pour point 0°F.

See Figure 2 for the temperature-viscosity curve for this oil.

Procedure

The load bar deflection was obtained from the load-deflection curve for the bearing pressure to be tested. The cam was rotated to give the maximum deflection of the lever bar. The adjusting nuts on the tie-rods were then set to give the desired deflection of the load bar. When adjustments were satisfactory the test shaft was rotated by its motor. Speed was increased until the dying out of the test lamp or zero deflection of the ammeter indicated that an oil film had been established. The shaft was allowed to run to insure constant conditions in the bearing, and then gradually slowed until breakdown of the oil film occurred. This was evident by the deflection of the ammeter or flickering or glowing of the test lamp. A steady glow, barely visible, was arbitrarily assumed to be the point of breakdown of the oil film. Test journal r.p.m. and temperature of the bearing was obtained for this condition. The procedure was repeated several times until journal r.p.m. had been satisfactorily checked.

After obtaining breakdown data for steady load condition, the cam operating motor was started, and the load applied intermittently. With each revolution of the cam the load was alternately applied and relieved. The

journal speed was increased until a film was established, and then gradually slowed as before until film breakdown occurred.

Data was taken for bearing pressures between 150 and 5.0 pounds per square inch of projected bearing area for steady and intermittent loading. Test runs were made to determine journal speeds at film breakdown points for various frequencies of loading. Also data was obtained in an effort to correlate bearing temperature and oil viscosity with journal speed for a constant loading.

Secondly, the present inquiry will proceed from within, to seek out the internal dynamics of the system, and to explore the relationships between the internal dynamics and the external environment. This will involve an examination of the internal structures and processes of the system, as well as an analysis of the way in which the system interacts with its external environment. The inquiry will also seek to identify the key factors that influence the internal dynamics of the system, and to explore the ways in which these factors can be manipulated to improve the system's performance. Finally, the inquiry will seek to identify the key challenges and opportunities that the system faces, and to develop recommendations for addressing these challenges and capitalizing on these opportunities.

Oil Viscosity vs. Journal Velocity for Steady Loads

It is well known to all who have observed heavy machinery start from rest and run slowly that frictional resistance is great. This is evident from the groans emanating from the bearings, and the jerky irregular motion of the moving parts. Under these conditions there is metal to metal contact and interlocking of the minute irregularities of the surfaces. Boundary friction is occurring. As speed is increased, the groans become fewer and less intense, and finally the movement is smooth, even, and noiseless. The viscous film of oil has been formed, perfect lubrication is obtained, and a viscous friction condition is the result. Relative movement of the two surfaces has built up an oil film due to viscous drag, until they are entirely separated. Under such conditions friction is that due to the resistance to shear offered by the lubricant. The value of this resistance depends upon

1. Viscosity of the lubricant
2. Relative speed of the surfaces
3. Area of the surfaces
4. Thickness of the oil film.

Since the friction of journals is practically independent of the load when the speed is sufficient to

maintain a pressure film between the two surfaces¹, it would be valuable to know the relation of these variables one to another in establishing and maintaining the oil film. An attempt was made to correlate the first two, namely, the viscosity of the lubricant and relative speed of the surfaces. Time did not permit an investigation of the effect of varying surface area and thickness of oil film. This might easily be done by using test bearings of various diameters, and varying radial clearance respectively. The results of the investigation are shown in Figure 1. A logarithmic plot of these data shows that journal speed varies inversely as (Viscosity Saybolt Universal) 1.15 or Journal r.p.m. = $(\frac{1}{\text{Viscosity}})^{1.15}$ for a constant steady load.

¹ Lubrication and Lubricants, p. 107. Archibutt and Deely.

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Results

The results obtained have been divided into two relationships shown graphically by the curves of figures 1, 2, and 2A.

1. Figures 2 and 2A represent the direct results of the basic investigation. In Figure 2A oil film breakdown pressures in pounds per square inch of projected bearing area are plotted against corresponding journal speeds in revolutions per minute. These points were taken with the machine operating at a cam frequency of 100 per minute, and at bearing temperatures varying from 60° to 70°F. This curve indicates that the load carrying capacity of an oil film within a bearing of the type used under conditions of steady loading is greater than that of an oil film under conditions of intermittent loading. It is noted that the increase in breakdown pressures versus journal speeds is a straight line relationship in each of the two types of loading. The curves being practically parallel show that the breakdown pressure under steady loading exceeds that of the intermittent loading by about 45 pounds per square inch. In Figure 2 the breakdown pressures in pounds per square inch of projected bearing area are plotted against the function " $\frac{W}{P}X$ ", the symbols of which have the following significance:

η - absolute viscosity in centipoises

N - journal speed in revolutions per minute

P - bearing pressure in pounds per square
inch of projected area

(The absolute viscosity in centipoises was obtained by use of curve 3A, the temperature of the bearing being assumed to be the same as the temperature of the oil film. The error incurred by this assumption is believed to have no effect on the comparative value of the results.) Figure 2 shows the relationship of the breakdown pressures for the two types of loading with the temperature factor corrected for. That is, it eliminates the possibility of distortion due to changes in viscosity resulting from the 10°F. variation in bearing temperature. This curve also shows the breakdown pressure to be greater for steady loading than for intermittent loading.

2. In Figure 1 viscosity, Saybolt Universal, and bearing temperatures in degrees Fahrenheit were plotted against journal velocity in feet per minute. This curve shows that increases in bearing temperatures indicate lower viscosities and require greater journal velocities to maintain the oil film under a constant load. (In this case, the load was 280 pounds per square inch of projected area)

Discussion of Results

1. The results indicated in Figures 2 and 2A are contrary to generally accepted theories as to how the breakdown pressures under steady and intermittent loading should compare. Archbutt and Deeley make the following statement, "In some cases the loads upon bearings are by no means constant, for the faces often alternately approach and recede from each other. When this is the case, and the alternation is very rapid, the bearing will carry a very great weight, for at each alternation the pressure is completely relieved, and the oil 'trapped' cannot be expelled during the short time the load rests on the bearing."¹

It is probable that in the case of intermittent loading the load as applied to the bearing became eccentric due to a slight angular movement of the loading mechanism in the wake of the lever bar. This would cause the oil film to break down more quickly than if it were applied through the center of the bearing, as in the case of steady loading which existed, more nearly duplicated the conditions found in many engineering uses of this type of loading, as in crank pin and wrist pin bearings.

¹See "Lubrication and Lubricants" by Archbutt and Deeley.

REVIEW BY MICHAEL LEE

and the South, and the industrial economy with its
new role of an economic and social provider of specialized
national institutions like public radio stations, museums,
galleries and universities that provide economic benefits
and assistance to a range of other fields. In this, the
University of the South and the Canadian system of
not-for-profit and public sector higher education are
not so very much unlike their North American
counterparts and the very significant role that
the educational system of the United States plays in
the United States and the Canadian educational
system and not very many not unlike the United States
in applying not so

surprisingly to some very similar circumstances. As the
Canadian model shows us, the bottom line is that the national
university system and the provincial systems should be
like the upper classes that were taken out of the universities at
the time of the 1960s and 1970s, that are now the ones that are
going to bring with it the control of the economy and industry
and not universities. Given that, probably the best
model to follow is to have privatized ones of which would be
something like, perhaps the one that is in

the direction of the ones that are mentioned and
perhaps

2. A study of Figure 1 shows that variations in the temperature of the oil film and consequent changes in its viscosity have a marked effect upon the carrying power of the film.

and subsequently that there is enough to make a
reasonable judgment that this will not be associated with
polymer softening. Finally, it can be seen clearly that the
softening will be to a minimum.

Recommendations

It is recommended that further investigations of the behavior of an oil film in a bearing under steady and intermittent loads be conducted. In connection therewith the following suggestions are submitted:

1. The rigidity of that part of the loading mechanism including the load bar, the rods, and strong back should be improved to prevent angular motion of these parts as the load is applied. This should eliminate excessive vibration and the possibility of eccentric loading.
2. Finer speed adjustment of both the journal and can operating motors should be secured. Also a wider range of speed control for the journal operating motor is necessary in order to investigate conditions of loading greater than 600 lbs. per sq. in.
3. The effect of frequency of loading upon the breakdown pressures is still undetermined and offers an interesting field for further research.
4. The effect of time of application of the load, and the use of various types of oils suggest other lines of investigation.
5. It would be interesting, if possible, to analyze the breaking down process within the film from the point

of first metal to metal contact until maximum breakdown has occurred, for the two conditions of loading, and to determine at what stage of the process scoring of the bearing or seizure is apt to result.

This analysis might be made by use of a milliammeter in the indicator circuit, providing delicate speed and load control have been secured and there is very little vibration in the apparatus. The bearing should be removed and inspected for scoring after each desired point of breakdown has been reached.

political parties like yours. I do not know what to do
at this moment. I would hope and not choose any
of the options you give me but I will have to make up

my mind by the 21st this evening

regarding what you do with us now

the long awaited question. I have not had time to

make up my mind but I will have to do so by the 21st

because we have to make a decision on the 22nd

regarding the 21st. I will have to make a decision on the 21st

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Acknowledgment

Acknowledgment is made primarily to the Navy Department, which through the Post-graduate School of the U. S. Naval Academy made available both the time and the funds for this research work, and which also supplied the two necessary motors.

We are greatly indebted to L. J. Bradford, Professor of Machine Design at The Pennsylvania State College for his many helpful suggestions.

We also wish to express gratitude to L. A. Doggett, Professor of Electrical Engineering; P. C. Stewart, Associate Professor in Charge of the Mechanical Engineering Laboratory; and to the Department of Metallurgy for their aid and cooperation.

Introduction

that each of these three had a recognizable, and the second, a more distinct, though less intense, character. The first was a sharp, clear, and rather rapid series of short, high-pitched, and somewhat irregular, notes, which sounded like the rapid, sharp, and irregular notes of a small bird, such as a sparrow or a titmouse, or like the notes of a small, sharp, and irregular insect, such as a fly or a wasp. The second was a series of short, sharp, and rapid notes, which sounded like the notes of a small, sharp, and irregular insect, such as a fly or a wasp. The third was a series of short, sharp, and rapid notes, which sounded like the notes of a small, sharp, and irregular insect, such as a fly or a wasp.

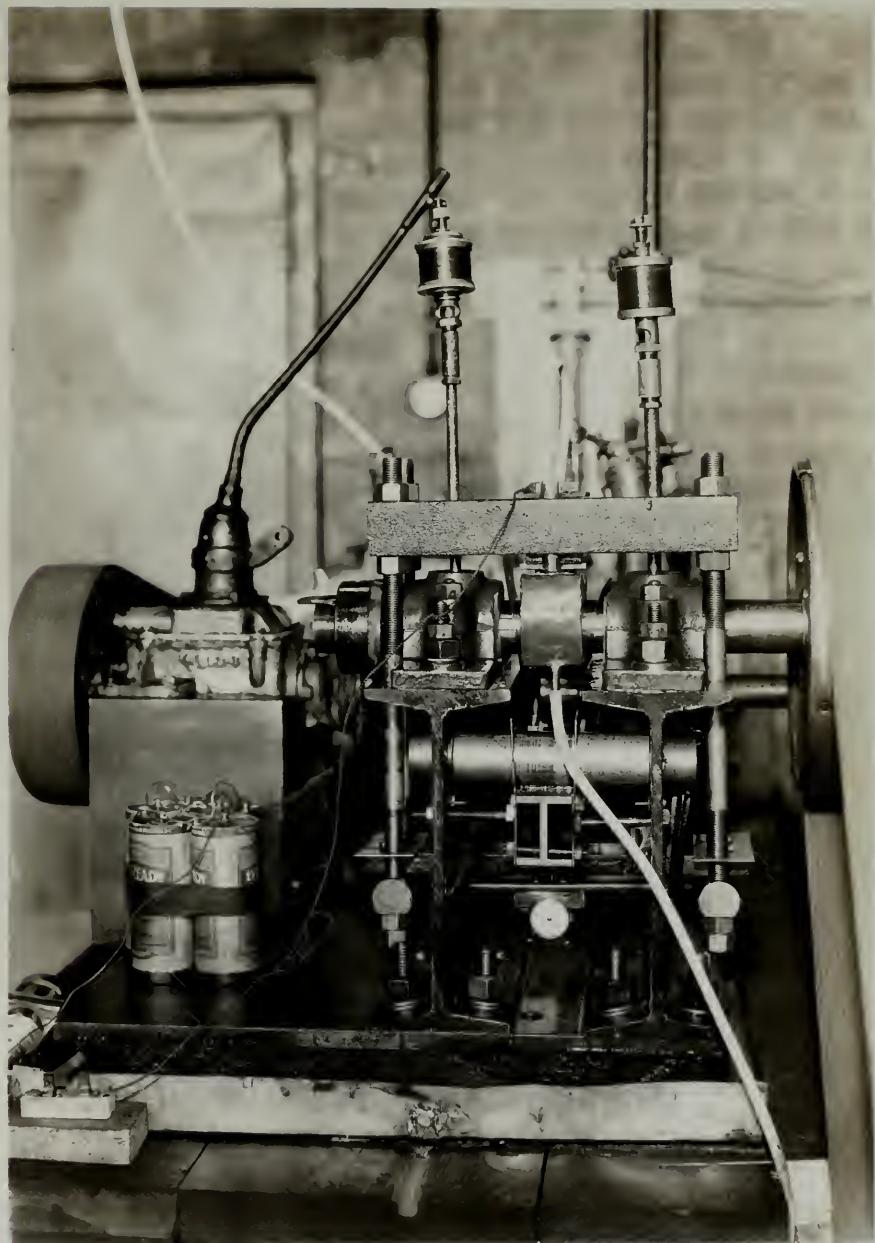
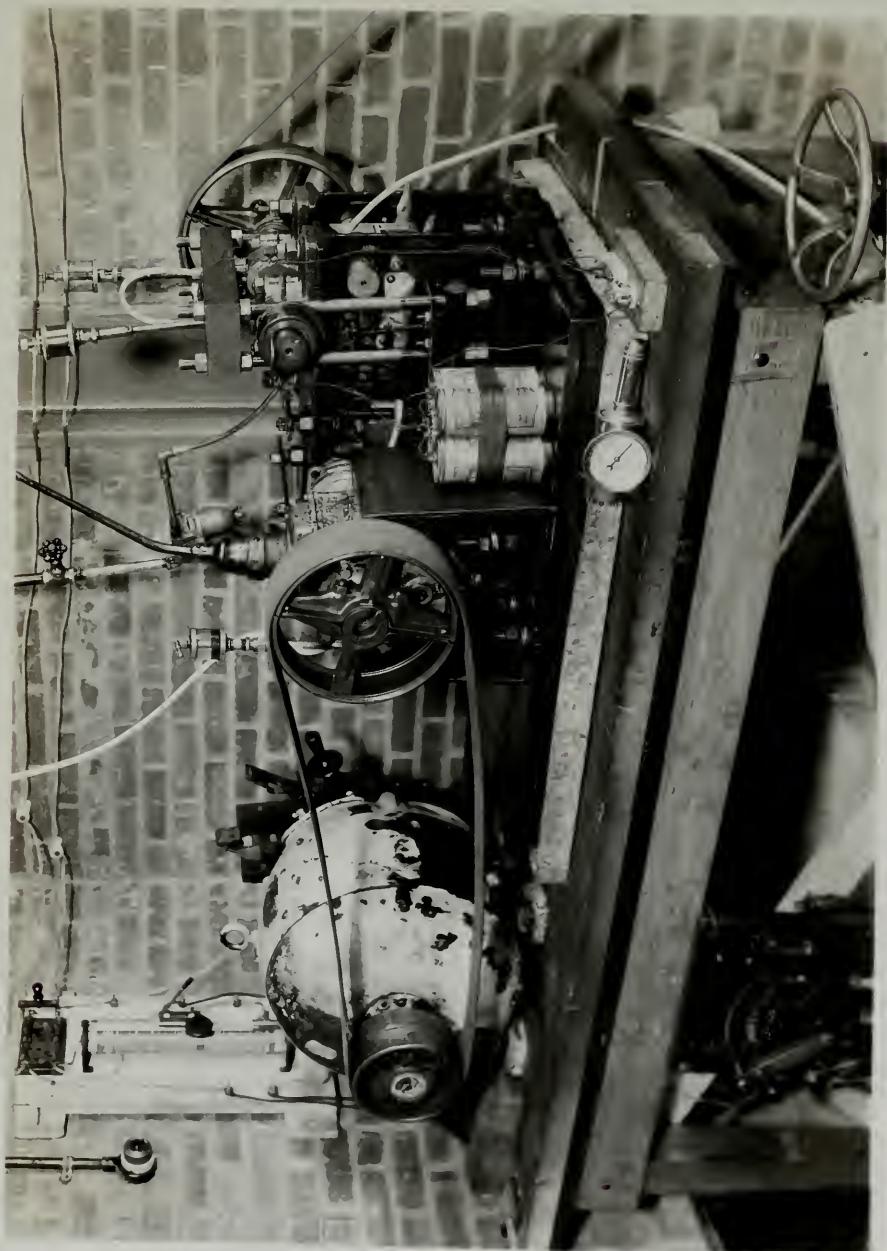


PLATE 1

END VIEW OF MACHINE

SIDE VIEW OF SICKLE AND CROWN CUTTING CAR

Plates 2

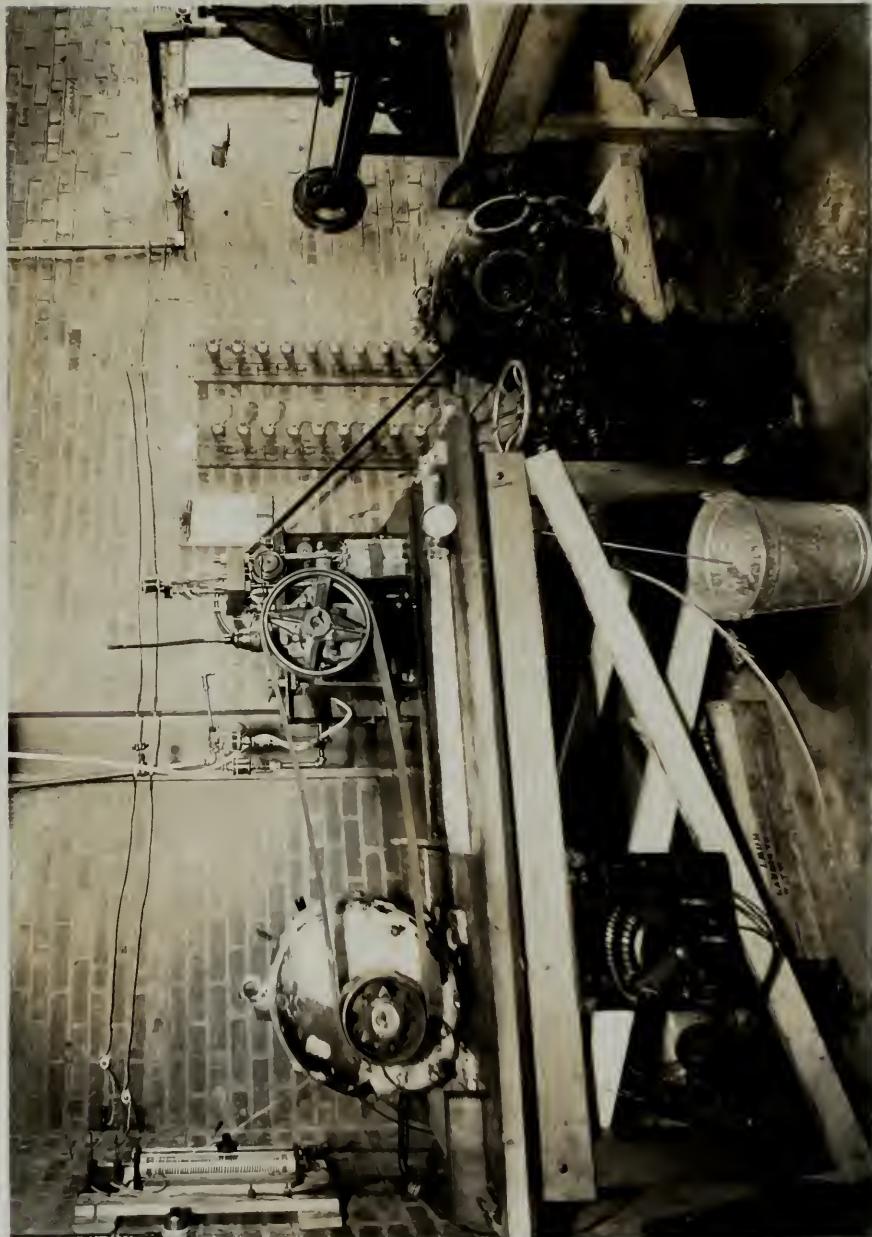


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PLATE 3

ASSEMBLY VIEW OF MACHINE, OPERATING AND C.G.D. CONTROL EQUIPMENT



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50000



SAYBOLT
UNIVERSAL TEMP
VISCOSITY °F.

150 90

170 80

1970 70

1750 60

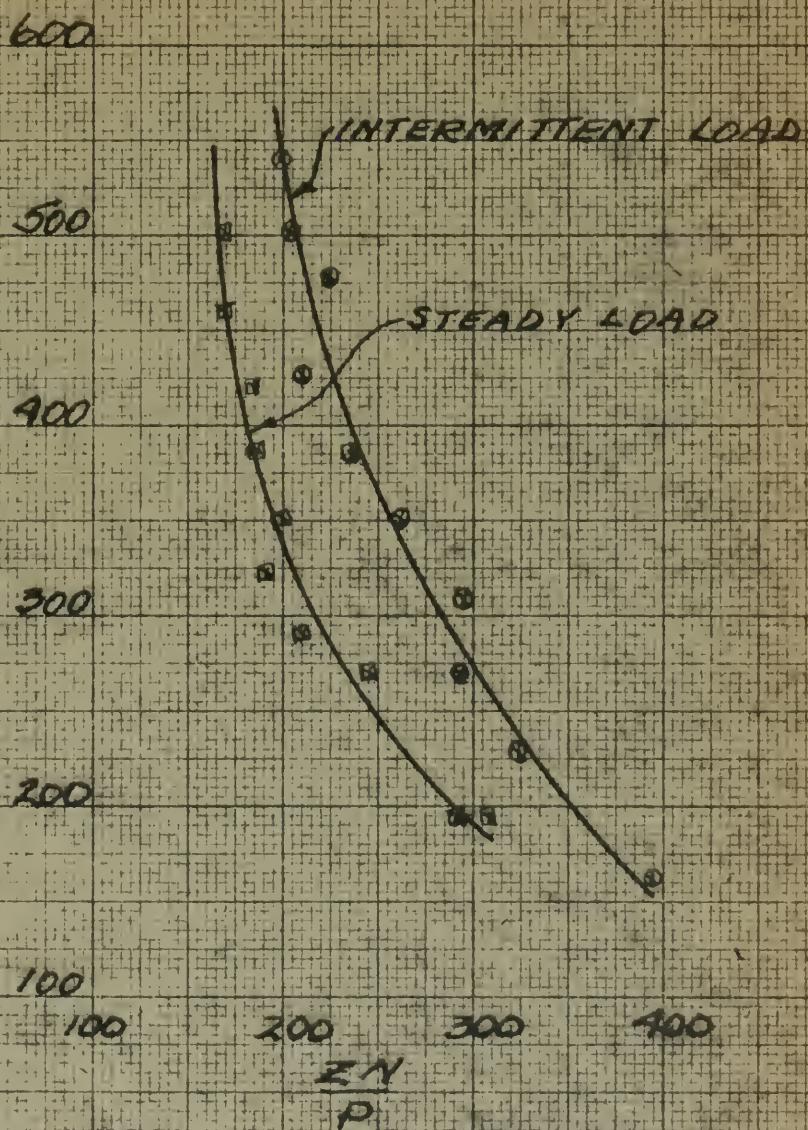
100 150 200 250 300

JOURNAL VELOCITY - FPM.

BEARING PRES $280^{\ast}/\text{in}^2$ OF PROJ AREA

FIG. 1.

BEARING
PRESSURE
1/2" OF
PROJECTED
AREA



~~Z = ABSOLUTE VISCOSITY IN CENTIPOISES
N = RPM OF JOURNAL
P = BEARING PRESSURE IN $\frac{lb}{in^2}$ OF
PROJECTED AREA.~~

FIG. 2

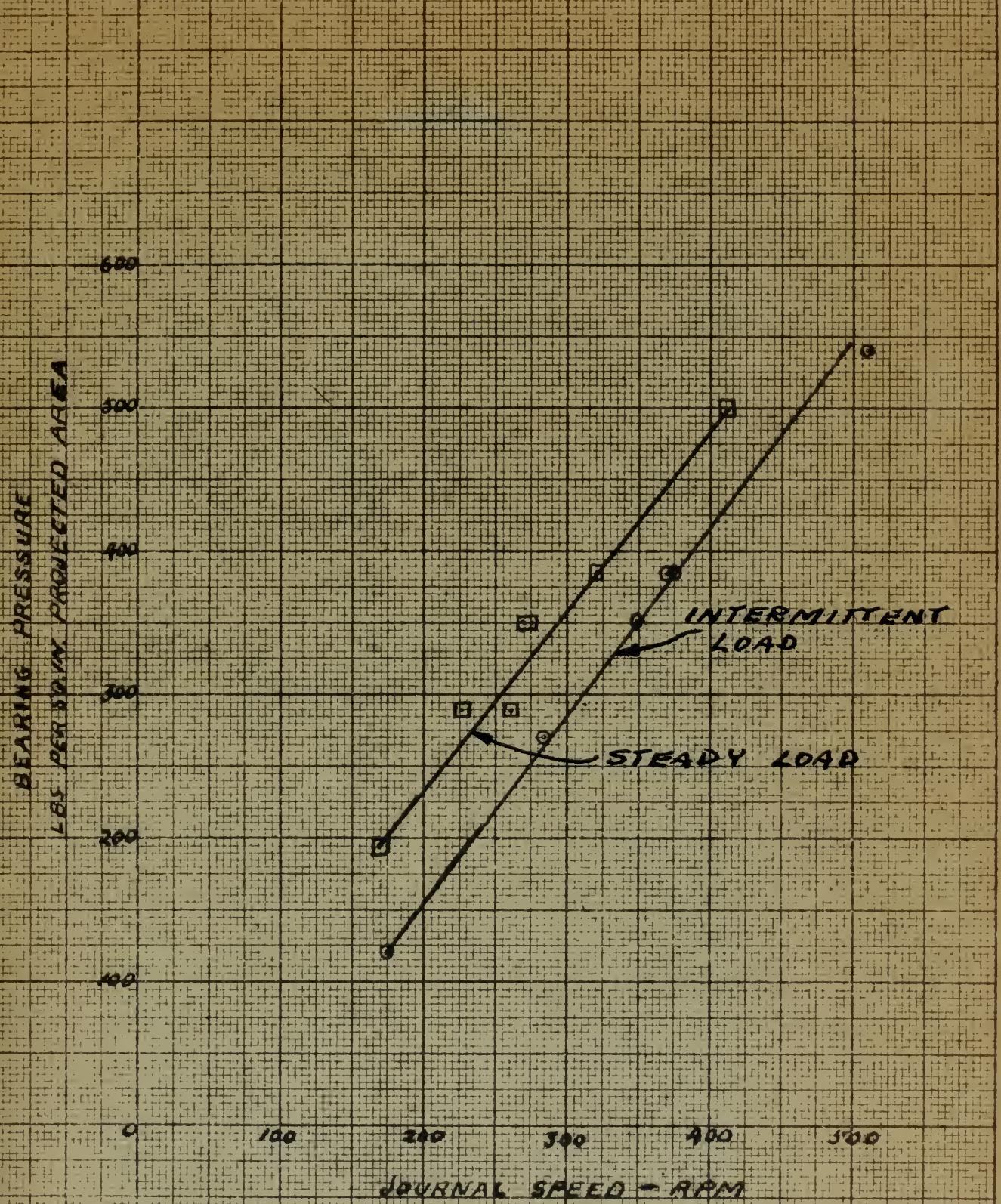


FIG 2a



VISCOSITY
SAYBOLT
UNIVERSAL

1800

TEMP - VISCOSITY CURVE

1600

FOR

1400

TEXACO REGAL "C" OIL

1200

1000

800

600

60

65

70

75

80

TEMP. °F.

FIG 3



TEMPERATURE-VISCOSITY
CURVE
FOR
TEXACO REGAL "C" OIL

ABSOLUTE
VISCOSITY
IN
CENTIPOISES

400

300

200

100

60 65 70 75 80

TEMP-OF

FIG. 3A

BEARING
PRESSURE

$\frac{\text{PSI}}{\text{IN.}^2}$
OF
PROJECTED
AREA.
1000

800

600

400

200

0

.005

.010

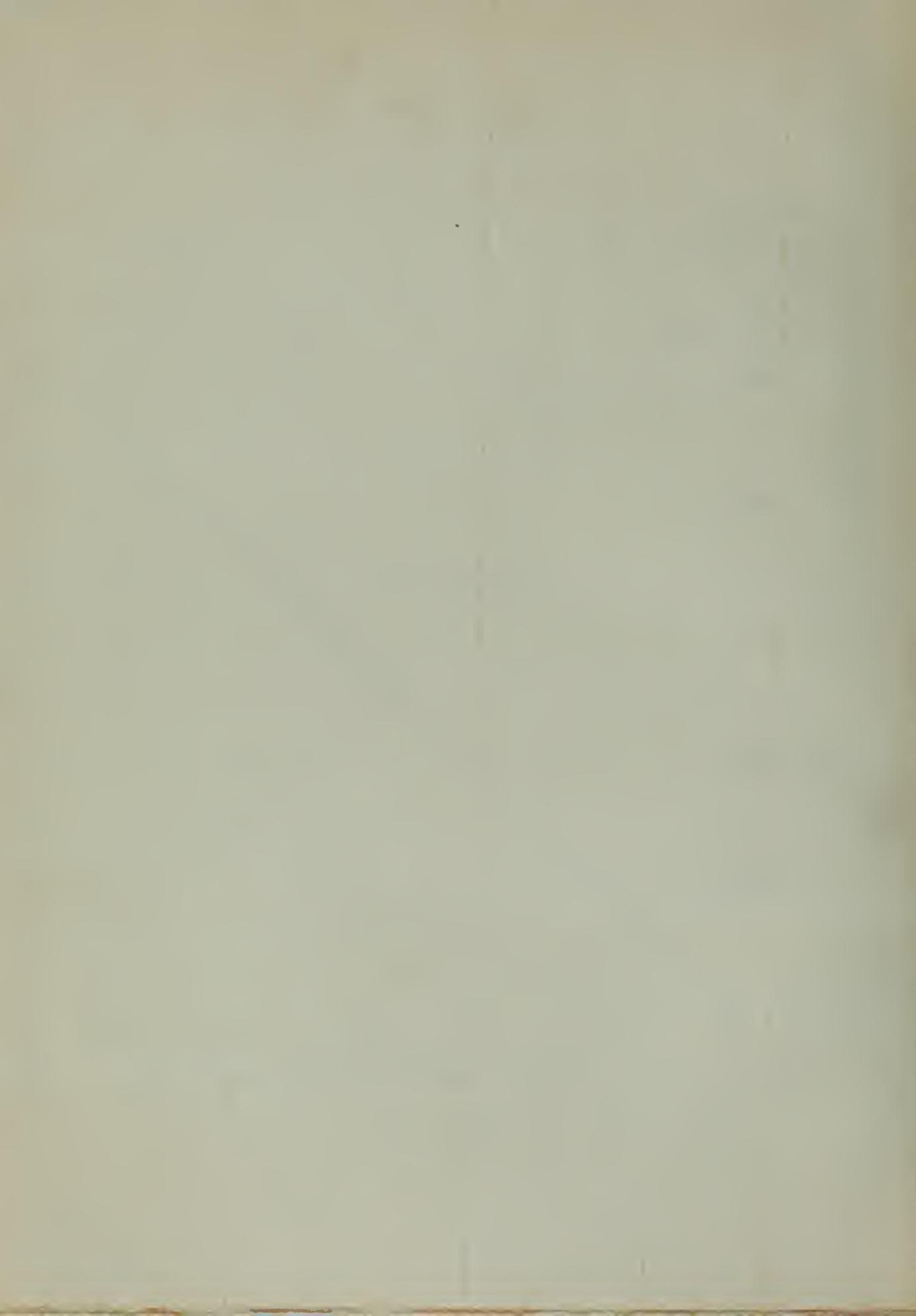
.015

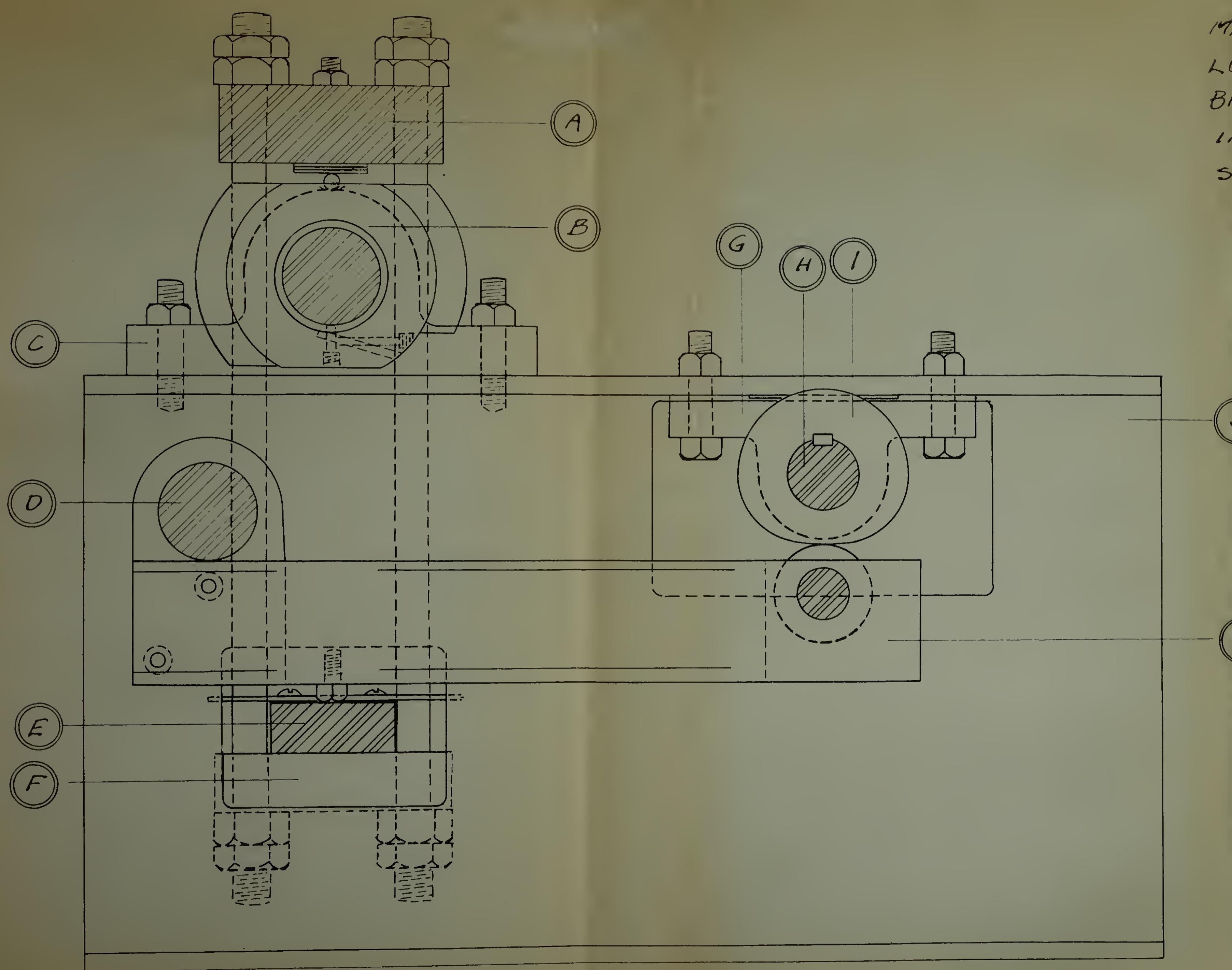
.020

.025

BEAM DEFLECTION - INCHES

FIG. 4





MECHANISM FOR MEASURING
LUBRICATING OIL FILM
BREAK-DOWN PRESSURES
IN A BEARING UNDER
STEADY AND INTERMITTENT
LOADING

A.D. BARNES.
D.N. CONE.

KEY TO DIAGRAM	
DES	PART
A	STRONG BACK
B	TEST BEARING
C	SUPPORT BEARINGS
D	FULCRUM SHAFT
E	LOAD BAR
F	LOAD BAR SUPPORT
G	CAM SHAFT BEARING
H	CAM SHAFT
I	LOADING CAM
J	FRAME I-BEAM
K	LEVER BAR

FIG. 5.

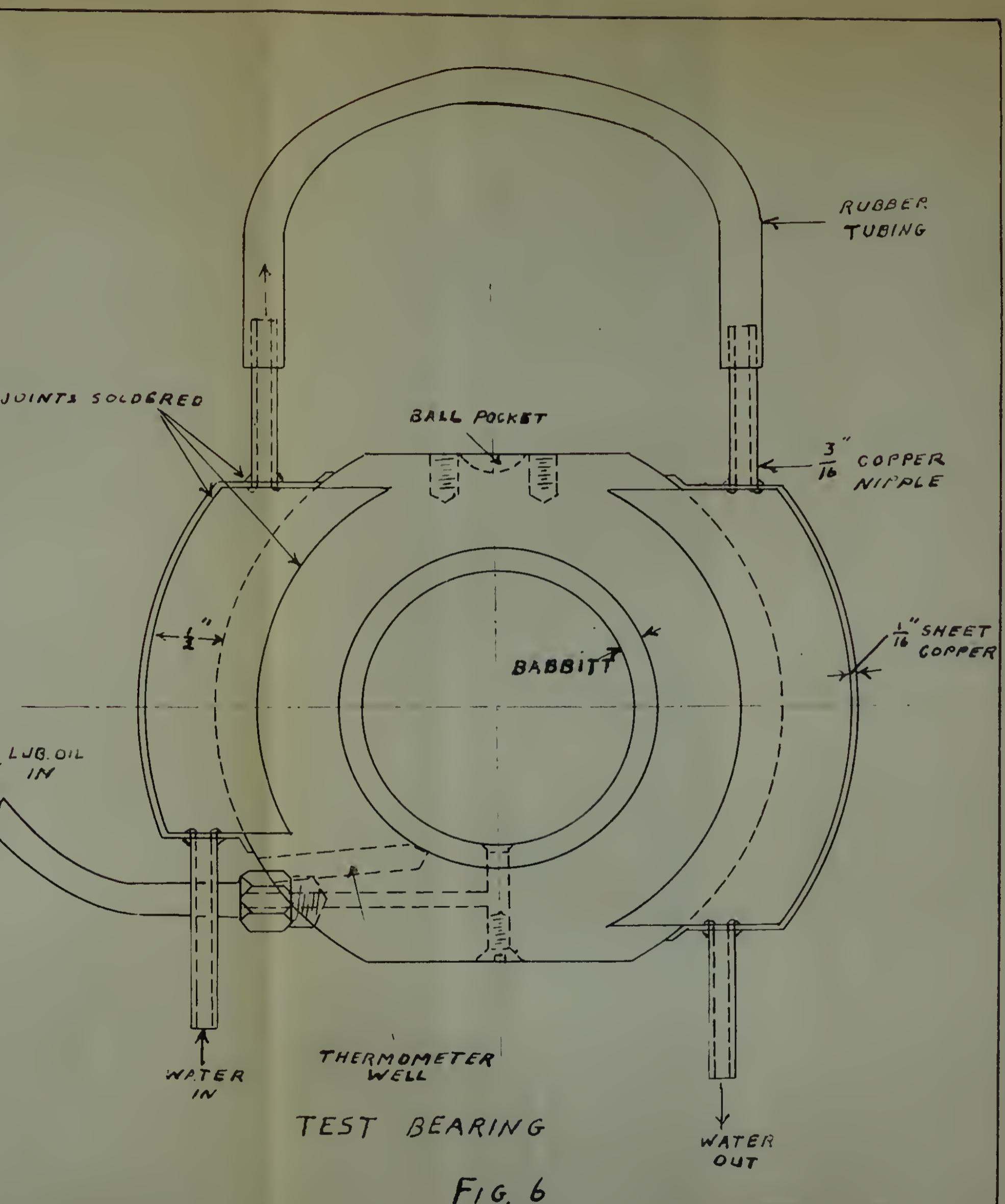


FIG. 6

AUG 31

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AUTHOR

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cating oil film ...

DATE LOANED	BORROWER'S NAME	DATE RETURNED
2	BROTHERTON, W. D. 4/11/52	NH 3 11
		of
		break-
		ready
		3.

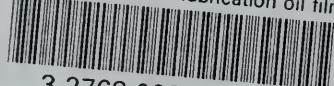
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pressures under steady and
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